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PORTABLE AIR COMPRESSORS: THE COSTS, BENEFITS, AND PENALTIES OF REDUCING THEIR NOISE

T. W. Freeze, et al

Bolt Beranek and Newman, Incorporated

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PORTABLE AIR COMPRESSORS: THE COSTS, BENEFITS, AND PENALTIES OF REDUCING THEIR NOISE

T.W. Freeze
H.L. Fox
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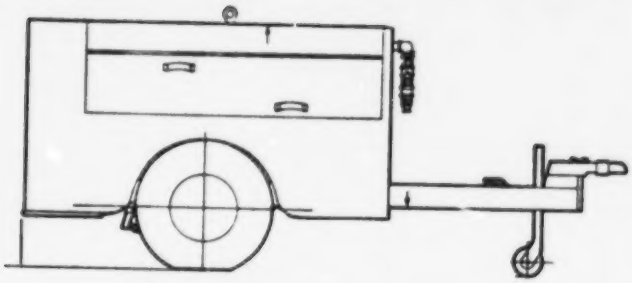
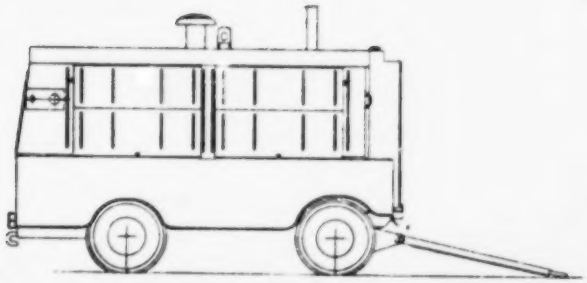


SMALL 2-WHEEL COMPRESSOR



LARGE 4-WHEEL COMPRESSOR

PORTABLE AIR COMPRESSORS.

ABSTRACT

Noise and list price data, obtained primarily from the portable compressor industry itself, indicate that about 50% of the gasoline-engine powered compressor models presently marketed as "quieted" ("silenced", "muted", etc.) are at least as quiet as 76 dB(A) at 7 m (per ISO 2151-1972) at a mean list price of $43.32 per cfm. Corresponding figures for diesel-engine driven units are 76 dB(A) at $52.11 per cfm for those delivering 500 cfm or less and 78 dB(A) at $48.70 per cfm for those delivering more than 500 cfm. Lower noise levels and higher prices are similarly determined for the "quietest" decile. Noise reduction to achieve the sound pressure levels of the present average quieted machine, the quietest decile, and levels below the quietest decile is accomplished for each regimen by the use of three overlapping technologies: sound insulation, sophisticated sound insulation and vibration isolation, and noise source elimination such as fan redesign and alternative engine designs. The data yield three corresponding regimens of marginal cost [$/cfm/dB(A)]. These are offered as the basis from which to project prices and determine market impact. Among other topics discussed are measurement standards, deterioration of noise control with use and aging, and on-site noise of compressors.

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1. INTRODUCTION

The noise generated by portable air compressors is to many people the most evident and the most annoying of the noise resulting from construction operations. Furthermore, many citizens associate the noise of compressors themselves with the noise of equipment powered by compressors. Wherever rock drills, jack-hammers, and pile drivers are operating, the drone of the air compressor provides the incessant background to the noise that pummels the ears of an increasingly noise-aware population. Construction firms, compressor manufacturers, and regulatory agencies have responded to the need to quiet portable air compressors and associated equipment. The result has been widespread activity, not all of which has been efficient or coordinated. For example, contractors and equipment manufacturers often do not know what standards their equipment must meet.

An important step toward bringing some order to the situation was the issuance in 1969 of a uniform test code for the measurement of sound from pneumatic equipment, prepared in cooperation by the Compressed Air Gas Institute (CAGI) and the European Committee of Manufacturers of Compressed Air Equipment (PNEUROP) [1]. By standardizing measurement procedures of portable air compressors, the CAGI-PNEUROP test code greatly facilitated the manufacturers' response to the increasing demand for quieter equipment. CAGI's attention to noise and the widespread establishment of committees on noise in the participating firms has resulted in a general knowledge of noise reduction technology and of basic acoustical concepts throughout the industry.

There is evidence that the increasing demand for quieter compressors has made the industry more competitive. Although sales for the leading two firms have increased, there is even greater growth among the lesser ranking firms. Consequently, there is a considerable scramble for leadership in selected categories of portable compressors among the eight firms that account for over 95% of sales. But this competition for a greater share of the market is not reflected in competitive pricing.* It is reflected more in service, reliability, and noise reduction. Those firms with the greatest increases in sales are generally the ones that are introducing more advanced noise reduction technology. The portable air compressor industry is thus very active, competitive, and noise conscious.

The development of quieter compressors is probably due not only to an increased demand for quieter equipment, but also to an overall increasing demand for compressors in general, which motivates manufacturers to invest in new design and capital equipment. Total sales of portable air compressors delivering over 75 cfm at 100 psi are likely to be close to $70 million in 1973, and were $58.9 million in 1972 and $49.4 million in 1971 [2]. As a result of this competition for sales in an expanding market that is very sensitive to noise problems, some firms about to introduce new lines or in the process of competing for large bids refused to give us data. Nevertheless, we obtained data from twelve firms of which we visited six. These six were from among the top eight in sales. Furthermore, the data cover 194 models which

*List prices for standard machines show scarcely any evidence of price competition and very little is evidence in the prices for the "quieted" units. Price cutting does occur to, especially in competitive bids for large procurements. In this report all price data were furnished by the manufacturers in response to unqualified requests for prices. We have subsequently been advised that they are list prices and may not represent actual market prices. See Appendix D for list price relations to pumping capacity.

we estimate to be well over 90% of those offered to the U.S. market. Our data base is well representative of the ten to eleven firms having sales over $100,000 in the three compressor size categories 75 to 150 cfm, 151 to 400 cfm, and over 400 cfm [2]. A list of the fifteen firms contacted is provided in Appendix A.

Because portable compressors have a reputation for loud noise and because price increases may be necessary to cover costs of noise reduction, the questions of import competition and substitute products are important. We therefore considered two non-U.S. manufacturers who supply the U.S. market and we examined the European noise control literature. There does not seem to be unusually severe competition from the imports, especially in consideration of the effect of dollar devaluation on price competitiveness. Furthermore, several U.S. manufacturers sell to European (and other) markets in which they must conform to stringent noise emission requirements.

Some of our sources considered the threat of substitution to be severe. They suggested that, in the long run, the threat of hydraulic tools to pneumatic tools was increased by the cost disadvantages imposed on compressors because of the requirement for noise reduction. We did not investigate this proposition because it was not widely emphasized, it represents a fairly long-term possibility, and the threat of substitution of hydraulic compressors is not from another sector so much as a potential realignment in the construction equipment industry.

2. AIR COMPRESSORS TODAY

Portable air compressors delivering over 75 cfm are sold primarily to the construction industry, utilities, municipalities, and for rentals. They generally are rated according to maximum flow rate of free (ambient pressures) air at a pressure of 100 psi (although some firms have units rated up to 150 psi). The maximum flow rate ranges as high as 2000 cfm. In 1972, 6,323 units rated at 75 to 150 cfm were sold at an average unit price of $2,445; 3,958 units rated at 151 to 400 cfm at an average unit price of $3,755; and 1,873 units rated above 400 cfm at an average unit price of $15,300 (see Table I).

TABLE I. QUANTITY AND VALUE OF SHIPMENTS IN 1971 AND 1972 IN THE U.S. OF PORTABLE AIR COMPRESSORS DELIVERING 75 CFM AND ABOVE

Compressor Range	No. of Companies with Shipments over $100,000	1972 Unit Quantity	1972 Value ($1000)	1971 Unit Quantity	1971 Value ($1000)
75 - 150 cfm	11	6,323	15,448	5,319	13,263
151 - 400 cfm	10	3,958	14,842	3,017	12,176
above 400 cfm	10	1,873	28,607	1,565	23,996
TOTAL		12,145	58,897	9,901	49,435

Almost all the larger units are diesel-engine driven, screw-type compressors; the intermediates are diesel and gasoline-engine driven, screw and rotary-type compressors; and the smaller types are primarily gasoline-engine driven, screw, rotary and reciprocating-type compressors (see Table II). For all types, the primary noise sources on units without noise reduction are the driving engine and the air system used to cool the engine and compressor (usually through intermediate oil and water systems).

TABLE II. DISTRIBUTION OF ENGINE TYPES AND COMPRESSOR DESIGN TYPES ACCORDING TO RATED ENGINE CAPACITY IN CFM AT 100 PSI

Compressor Type	75-200 cfm			201-500 cfm			Above 500 cfm		
	Gasoline	Diesel	Gasoline and Diesel	Gasoline	Diesel	Gasoline and Diesel	Gasoline	Diesel	Gasoline and Diesel
Reciprocating	16.6%	10.3%	26.9%	0%	30.8%	30.8%	0%	6.8%	6.8%
Vane	25.6%	19.2%	44.8%	10.3%	33.3%	43.6%	0%	17%	17%
Screw	15.4%	12.8%	29.2%	2.6%	23.1%	25.7%	0%	76.3%	76.3%
All types	57.6%	42.3%	99.9%	12.9%	87.2%	100.1%	0%	100.1%	100.1%

As increased quieting is attempted, the compressor and the associated valving, air intake, and mechanical vibration become the necessary objects of any noise control.

2.1 The Basic Elements of Portable Compressor Equipment

The portable compressors with which we are concerned are designed to be towed as trailers on two or four rubber-tired wheels. They have weights ranging from 1 to 14 tons, lengths from 6-1/2 to 19 ft, and heights from a little less than 6 to almost 10 ft. On the trailer is mounted the compressor, an air receiver, driving engine, cooling system, fuel tanks, tool boxes, and an enclosure. The enclosure itself when designed for noise insulation can weigh as much as 10% of the total weight.

2.1.1 Rotary screw compressors

Today, the most widely manufactured compressor type in the U.S. is the rotary screw. An axial pump that achieves very high

flow rates in a single stage, it provides a high flow-to-size ratio and, because it has few moving parts, is reliable. Typically, rotary screw units used in portable compressors are smaller in size than an automobile automatic transmission.

The basic operation is illustrated in Fig. 1. As the end view snows, the essential moving parts are a female and male helical rotor having mating grooves and lobes respectively. A port at one end of the housing admits air which is then captured in a groove and compressed as the corresponding lobe reduces the volume between itself and the far end of the housing. A port at the far end provides an outlet for the compressed air. Oil under pressure that seals and lubricates the rotor mating surfaces experiences a considerable temperature rise due to the heat of compression; hence, a substantial oil cooling system is needed. To drive the compressor, an engine occupying 5 to 15 times the volume occupied by the basic compressor itself is needed along with the accompanying cooling exhaust system. In

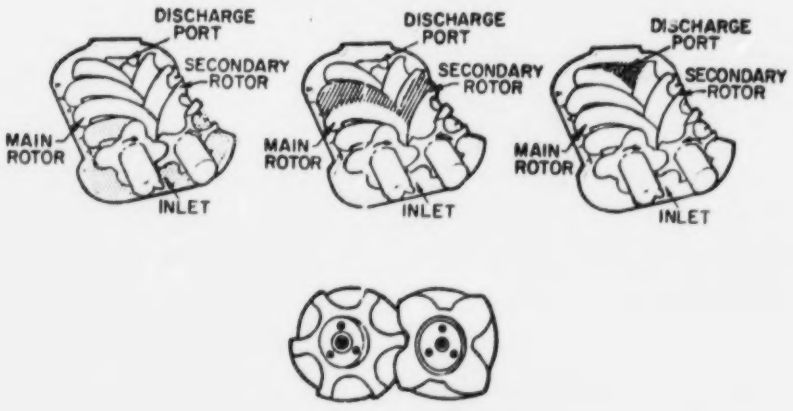

FIG. 1. ROTARY SCREW COMPRESSOR

most cases, the engine is directly coupled to the male screw which then drives the female. It is clear then that the basic screw-type compressor unit accounts for only a small fraction of the weight and size of an operating portable compressor. Likewise, the compressor mechanism itself produces little of the noise generated during operations.

2.1.2 Sliding vane rotary compressors

Most U.S. manufacturers are phasing out their line of sliding vane rotary compressors, probably because they are reputed to require more maintenance and are less economical to operate than other types in use. Nevertheless, there are still several portable compressor sets of this type on the market.

The basic design is illustrated in Fig. 2. An eccentrically mounted rotor is equipped with sliding vanes. As the vanes pass the inlet opening, air is trapped. Because of the eccentricity, further rotation reduces the volume compressing the trapped air. Discharge ports are located where the compression achieved exceeds the delivery pressure. As in the case of screw type compressors, the compressor itself is relatively small but the necessary concomitant equipment is substantial. Sometimes the compressor itself is mounted in the receiving tank in order to save space. In such cases, treating the tank with sound insulation isolates the noise of the basic

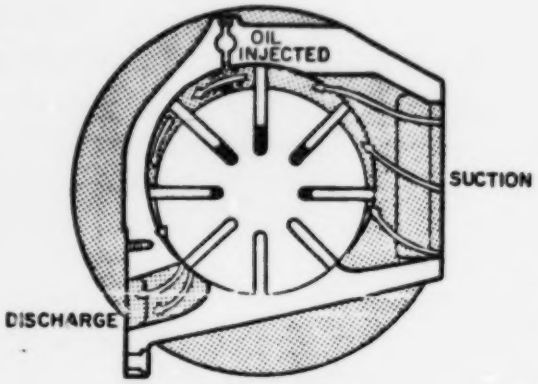

FIG. 2. SLIDING VANE ROTARY COMPRESSOR.

compressor unit. However, noise reduction is still required for the oil cooling system, engine, and other sources.

2.1.3 Reciprocating compressors

The traditional reciprocating compressor is today used almost exclusively in portable compressors delivering less than 250 cfm. Unlike the screw and rotary vane types, it usually requires several stages to achieve the required pressure. Consequently, the basic unit is a larger fraction of the total weight and size of the complete compressor. A simple understanding of its operation can be obtained by considering the compressor to be a reciprocating engine driven by its output shaft. Thus, all control is obtained by valving. In many designs the driving engine cylinders and the compressor cylinders are bored in the same block and connected through a common crank shaft. This arrangement makes it possible to use a single cooling system (usually water). The unitary construction also means that the compressor and engine are treated as a single problem with respect to noise reduction.

2.1.4 Engine, exhaust, and cooling systems for compressors

Engine drives for portable compressors delivering 75 cfm or more range from 2-cylinder gasoline or diesel engines to 16-cylinder diesels and from 30 to 650 hp (see Fig. 3). Mean operating speeds are about 2000 rpm under full load, but some engines run as slow as 1300 rpm and some as fast as 3600 rpm. Almost all units have direct drive from engine to compressor but a few have gear transmissions with step-up ratios as high as 4:1. Very few of the engines are made by the compressor manufacturers or made to specifications of the compressor manufacturers. Major suppliers of stock gasoline engines are Ford, Hercules, and International Harvester; major suppliers of diesel

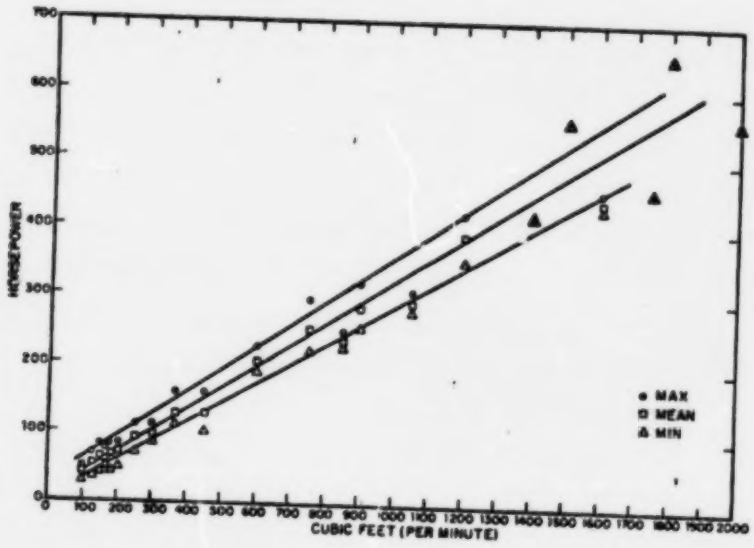

FIG. 3. POWER REQUIREMENTS OF PORTABLE COMPRESSOR. BASED ON DATA OF REF. 3.

engines are Detroit Diesel (GM), Cummins, Deutz, Ford, Hercules, and Caterpillar. These engines are generally noisy and require substantial cooling systems which also tend to be noisy.

Engine and compressor cooling in portable compressors requires very large flows of air. Estimates of cooling requirements for the larger units are as high as 15,000 Btu/min, requiring air/fan systems capable of moving 23,000 cfm of cooling air. Adequate cooling on large units surrounded by acoustic enclosures can require up to 30 to 40 hp to drive the fans. Furthermore, some enclosure designs, because they reduce cooling air inlet cross-sectional area, develop considerable flow velocities. Also, flow resistance introduced in acoustically designed ducts establishes static pressure rises of up to 3 in. of water.

A typical cooling system consists of two adjacent heat exchangers, one each for oil and water, and a fan and ducting

system that either draws the air over the engine and compressor and then through the heat exchangers or, in the more conventional reverse pattern, through the exchangers first. The heated air is emitted through vents in the roof, through the bottom, or at one end of the compressor trailer. Clearly, cooling systems pose considerable design problems and are important sources of noise.

Exhaust mufflers supplied with the engine do not usually reduce the noise sufficiently. Protruding from the top of the compressor 6 to 10 ft above the ground, they radiate the characteristic sound that most people associate with compressors. Fortunately, muffler technology is fairly well developed and in instances where the engine manufacturers supply the muffler, compressor manufacturers generally replace it with an available quieter one. But in many instances special designs are required.

2.2 How Compressor Noise is Measured

Most measurements of the sound emitted by portable compressors are made according to the standard originated by CAGI/PNEUROP and now adopted as ANSI S5.1 1971 and ISO 2151-1972. Relevant excerpts are in Appendix B. This standard provides a measure of the average and maximum sound pressure level at 1 m and 7 m from the equipment. The maximum level at 1 m corresponds closely to the measurement required by Section 1.403.3-5.11 of the New York City code. To satisfy codes for other areas that imply different measurements, manufacturers may make additional measurements. The most common is a measurement at 50 ft (~15.5 m), which corresponds to the distance at which the Chicago noise ordinance (adapted by several major cities) imposes noise level limits. Fifty feet is also the distance indicated in SAE Standard J952b and the GSA regulation on construction equipment procurements.

Measurements are usually made in an asphalt parking lot at the manufacturer's plant, at times of day when the background noise is low. The compressor is run at maximum rated output with all panels shut.

2.3 Noise of Present-Day Compressors

Examination of the noise emission of present-day compressors suggests that dividing compressors into six categories provides the most meaningful basis for evaluation. One division is into types of drive: gasoline *vs* diesel engines. A second is into "standard units" *vs* those offered as "quieted units". The two alternatives for the two characteristics, gasoline *vs* diesel and "standard" *vs* "quieted", define four categories. In each of these categories, price is a linear function of cfm rating (as would be expected where there is always the option of substituting two smaller units of half the rating for one larger unit).

The noise levels of standard diesel driven units rated above 500 cfm are bracketed above the noise levels of units rated at less than 500 cfm. Also, the corresponding quieted units have different ratios of price to cfm. Consequently, we divide the data for diesel engines as shown in the column headings of Table III.

For each category Table III provides the mean and standard deviation of price/cfm and sound levels at 7 m (measured according to ISO 2151-1972). It also indicates the sound pressure level of the quietest machine and the deviation of its price from the mean. Table III thus constitutes a summary of the present state of the noise and price of portable compressors being manufacture in the U.S.

TABLE III. PRESENT STATUS OF PORTABLE COMPRESSORS WITH RESPECT TO NOISE EMISSIONS AND PRICE PER RATED CFM

	Gasoline Driven		Diesel Driven			
			Below 501 cfm		Above 500 cfm	
	Standard	Quieted	Standard	Quieted	Standard	Quieted
Number of Units in Samples	32	26	45	35	32	24
Price/cfm						
Mean	$39.23	$43.32	$46.16	$52.11	$43.57	$48.70
Standard deviation	$ 4.40	$ 6.10	$ 4.57	$ 8.30	$ 3.56	$ 3.16
SPL at 7m						
Mean	82.8 dBA	76.1 dBA	86.1 dBA	76.4 dBA	92.8 dBA	78.7 dBA
Standard deviation	4.92 dBA	2.40 dBA	3.35 dBA	4.07 dBA	4.08 dBA	3.90 dBA
Quietest Machines (Lowest decile)						
No. in decile	3	3	6	4	4	2
Mean SPL at 7m	72.6 dBA	72.3 dBA	82 dBA	70 dBA	87.5 dBA	73.5 dBA
Deviation of average price in lowest decile from mean price of quieted	+$5.42	+$5.14	+$0.43	+$10.23	+$0.31	+$2.50

A 10.2 dB mean difference between standard and "quieted" compressors is offered at a mean price difference of $5.05 per cfm. Of particular interest is the fact that in the "standard" categories the quietest machines are priced on the average at only $2.05 above the mean price whereas the quietest of the "quieted" machines is on the average 9.1 dB quieter than the quietest "standard" machine but is priced about $5.96 above the mean price of the "quieted" machines.

In Table IV we show how present-day compressors compare with the Chicago and New York codes. In order to make this comparison it is necessary to extrapolate the Chicago and New York codes to the 7 m average SPL.

TABLE IV. PERCENTAGE OF COMPRESSOR MODELS THAT MEET OR ARE BELOW CHICAGO AND NEW YORK CITY CODES

Code	Gasoline Driven		Diesel Driven				Extrapolated Noise Limits at 7m per ISO 2151 1972
			Below 501 cfm		Above 500 cfm		
	Standard	Quieted	Standard	Quieted	Standard	Quieted	
Chicago 1973	97%	100%	87%	100%	39%	100%	91 dB(A)
Chicago 1975	91%	100%	75%	100%	18%	100%	89 dB(A)
Chicago 1980	34%	100%	21%	89%	0%	79%	83 dB(A)
New York 1973	9%	27%	0%	22%	0%	7%	74.7 dB(A)
New York 1974	0%	0%	0%	0%	0%	0%	64.7 dB(A)
New York 1975	0%	0%	0%	0%	0%	0%	59.7 dB(A)

The Chicago code applies to the highest sound pressure level reading obtained from four measurements taken at 50 ft from the sides of the compressor. The average difference between the maximum and average levels at 7 m is 2 dB according to the data we have. The attenuation due to spherical spreading from 7 m to 15.5 m (50 ft) would be 6.9 dB, but the data indicate an average of 5 dB. This deviation may be explained by the presence of the ground and the fact that 50 ft is still in the Fresnel zone of the compressor as a sound source. These corrections combine (in opposition) to yield an extrapolated noise limit of an average sound pressure level at 7 m that is 3 dB higher than the maximum allowed at 50 ft by the Chicago code.

To extrapolate the New York code, we must compare the maximum sound pressure level at 1 m with the average at 7 m. Again, the data do not show spherical spreading (16.9 dB). The average attenuation from 1 m to 7 m is, according to our data, 8.3 dB. The peak at 1 m is on the average 2 dB above the average at 1 m. These combine to yield an extrapolated noise limit at 7 m that is 10.3 dB less than the maximum allowed at 1 m by the New York code.

Table IV indicates that almost all presently manufactured "quiet" models meet the Chicago code through 1980. Few meet the current requirements of the New York code and none can meet its requirements for next year.

This situation is illustrated in Fig. 4, where we have plotted cumulative distributions. These display the percent of presently offered models that can satisfy any noise limit. The extrapolated noise limits of the Chicago and New York codes are indicated.

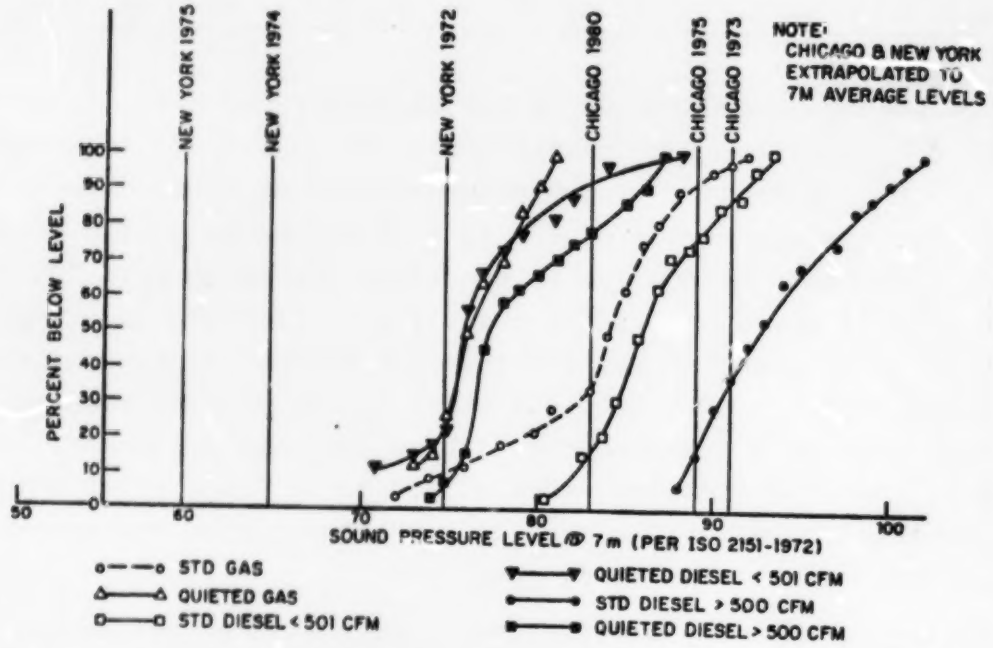

FIG. 4. CUMULATIVE DISTRIBUTIONS FOR EACH CATEGORY OF COMPRESSOR DISPLAYING PERCENT OF PRESENTLY OFFERED MODELS THAT CAN SATISFY A GIVEN NOISE LIMIT.

2.4 The Noise of Compressors on the Job

The data given in the previous section are quite sterile, since they represent standardized measurements rather than on-the-job performance. The actual annoyance, perceived noise, and sound pressure levels depend on a variety of factors related to the environment in which the compressor is operated, such as location of nearby noisy sources, position of observers, operator-adjustable features of the compressor, and age or state of repair of the equipment.

In city streets there may be building walls near the compressor. These walls will distort the pattern of sound radiation around the machine and greatly increase the peak-to-average ratio for an observer walking around the machine. For example, there is likely to be an increase of a few dB over the standardized measurement for an observer walking on a sidewalk between a compressor on one side and a building wall on the other. Also, the actual sound contour around the machine may have a larger peak-to-average ratio than is obtained in some application of ISO 2151-1972. We observed several test data in which maximum referred to one of the four measurements at the cardinal points. Figure 5 illustrates sound contours obtained by walk-around measurements. Note that the actual maximum is not at one of the cardinal points.

A more serious problem is the noise heard by an observer in a building above the street where the compressor is operating. The standard provides for no measurement above the machine but our observations and manufacturers' testimony suggest that there is considerable acoustic radiation off the top. Two sources are evident: many manufacturers mount the muffler on the top and several of the "quieted" models have vents on top.

Standard measurements are taken at full-rated output. Comparative measurements at idling indicate that the full-rated output is always the noisiest condition, but other conditions of the standard measurements are not replicated on-site. It is common to see portable compressors operated with the doors open or not locked tight. On the larger sound-insulated units this can increase the sound radiated by several decibels.

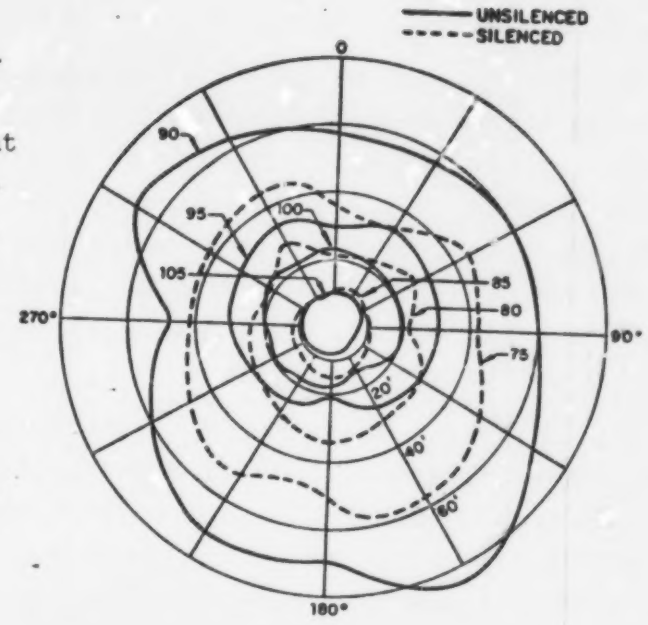

FIG. 5. OVERALL NOISE (A-SCALE) CONTOUR LEVELS, COMPARISON OF SILENCED AND UNSILENCED MACHINES (FROM REF. 4).

Noise insulation can also deteriorate, especially seals on doors. Also, loose mechanical parts tend to rattle as they wear. As a consequence, it is not uncommon that a quieted compressor after a year or two of heavy use has increased noise emission of from 4 to 8 dB.

Compressor noise usually has a spectrum that peaks at a frequency corresponding to the engine firing frequency. This gives it its annoying drone character. Octave band spectra are displayed in Fig. 6.

In summary, the compressor on the job tends to be more annoying than would be indicated by a standard test. Part of

the annoyance is traceable to actual sound pressure levels that exceed those measured under the idealized conditions of the test.

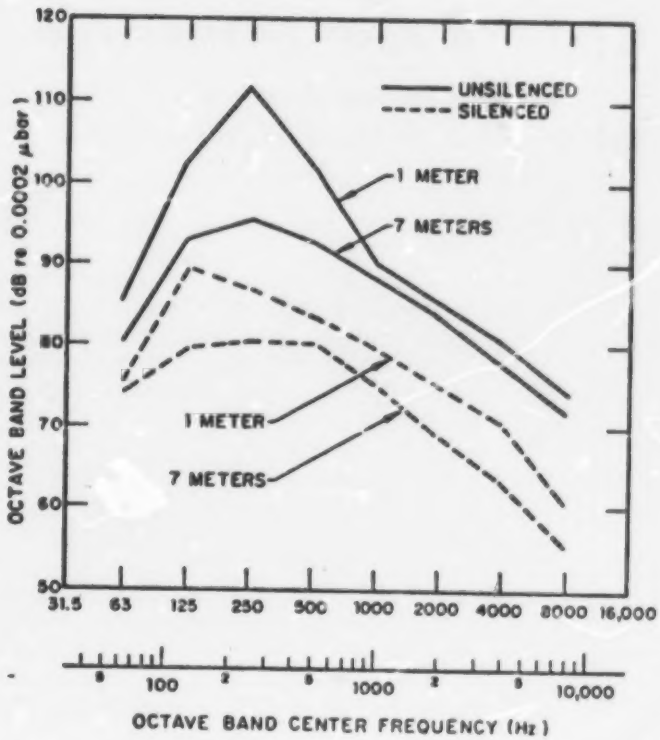

FIG. 6. COMPARISON OF NOISE OF SILENCED AND UNSILENCED COMPRESSORS (FROM REF. 4).

3. THE TECHNOLOGY AND ECONOMIES OF QUIETING AIR COMPRESSORS

With very few exceptions, the portable air compressor industry has reduced noise emission from its products by means of acoustic insulation. In the words of an important contributo to existing noise insulation technology in the industry [4]:

> One approach to noise reduction would be to redesign the equipment in such a way that the individual components were less efficient producers of noise. While this is possible in theory, it may not be possible in practice due to the effects such design changes might have upon the size, power level, and overall cost of the apparatus. In addition, the length of time required for successful completion of all such source reduction developments would probably be inconsistent with the current demands from the public. The alternative approach ... [is] to apply known techniques of muffling and enclosure to obtain the necessary overall noise reduction.

Unfortunately, the latter approach, while providing dramatic effects for the first 10 to 20 dB (at a considerable weight and heat-rise penalty), can offer little hope of achieving the reduction that will ultimately be necessary to respond to an increasingly noise-aware population. Conversion to and from acoustical energy is inefficient. An approach that requires the *reconversion* of acoustic energy to thermal or mechanical energy is less effective than *preventing* the conversion of mechanical, thermal, or aerodynamic energy to acoustic radiation in the first place.

There are other consequences of this approach. By relegating the problem of reducing noise generation to some undetermined future, the major manufacturers have created a vested interest (capital, inventory, development) in their present "quiet models". They react to the possibility of stringent regulations as unacceptable cost burdens. Their estimates of what can be done

at any cost is based on their experience with sound insulation and this leads to a very steep noise reduction/cost estimate for further noise reduction. However, the new firms, just beginning their investment, looking to an increasing demand, and aware of the relative inertia of the big firms, find themselves with a potential advantage: they can design their new lines with noise reduction built in. [E.g., we have been told of experimental units producing noise of 68 dB(A) at 1 m and observed one producing 73.5 dB(A) at 1 m.], This is, however, only a *potential* advantage, because the otherwise very able and inventive product and design engineers in the compressor industry do not generally have the sophistication in acoustical science necessary to integrate the task of source-noise removal with compressor design.

Another consequence is that the compressor industry has accepted the task and imposed the cost on their customers (through price differentials) of quieting noisy components supplied by producers of intermediate goods, primarily engines and fans. Had noise reduction been integrated in the product design from the start, present awareness of the engine's role in noise production might have stirred the industry to establish a uniform noise code for engine suppliers. Of course, the compressor manufacturers do not alone control the demand for engines and could not have imposed such a code. However, if other buyers of these engines (such as truck and tractor manufacturers) could have been stimulated to follow suit, the combined market pressure on the engine manufacturers might then have been sufficient to force them to produce quiet engines and air cooling systems.

An alternative way of dealing with the problem of noisy intermediate goods is for a regulatory agency to impose noise limits on these goods. To allow engine manufacturers to produce

noisy engines while imposing noise limits on construction equipment manufacturers who use the engines is to accept noise pollution as an externality to the engine manufacturing sector while compelling the construction equipment industry, construction industry, and consumers of construction services to bear the cost. This is not only an obviously unfair transfer of the burden, it is also wasteful. First, the costs of corrective measures are always more expensive in the long run than the costs of producing equipment that is designed to eliminate noise generating mechanisms. Second, there is considerable duplication of effort among the different industries in the construction equipment sector and among firms within each industry.

Awareness of the extensive duplication of effort led several of our interviewees to suggest a cooperative effort of the compressor industry as a whole, a government agency, and a research and development firm to produce the needed noise elimination and reduction technology for all firms in the industry. Imposing noise limits on the products of the engine supplying industry would also contribute towards eliminating wasteful duplication of effort.

The above considerations structure the evaluation of the technology and economies of quieting portable air compressors. One must consider two technologies. The current technology - primarily quieting by acoustic insulation - which the current economic structure reflects, yields practical limits to noise reduction not much lower than now achieved. A very few firms are experimenting with the technology of noise source elimination which could yield noise reduction to meet the most stringent currently projected codes. The economic analysis for this technology must be more speculative because there is presently insufficient data.

3.1 Quieting by Acoustic Insulation

Acoustic insulation, quieter mufflers, shock mounts, damping, and some fan, cowling, and duct improvements generally describe the technology that makes some of the presently offered portable air compressors quieter than others. In particular, the differences in sound pressure level between the "standard", "quieted", and "quietest" units given in Table III (see Sec. 2.3) are due to this technology. Figure 7 summarizes the noise/price contours given by the data of Table III. To generate the curve for each category, we use three data points corresponding to the mean sound pressure level and mean price of the "standard", "quieted", and "quietest" decile of the quieted units.

Most of our interviewees contended that costs of quieting equal or exceed the corresponding price differential. If additional costs equal price increments of quieted over standard compressors (on the average), the marginal cost data of Table V can be derived.

The simple assumption used to construct Table V – incremental costs equal incremental prices – is not entirely satisfactory. We have observed

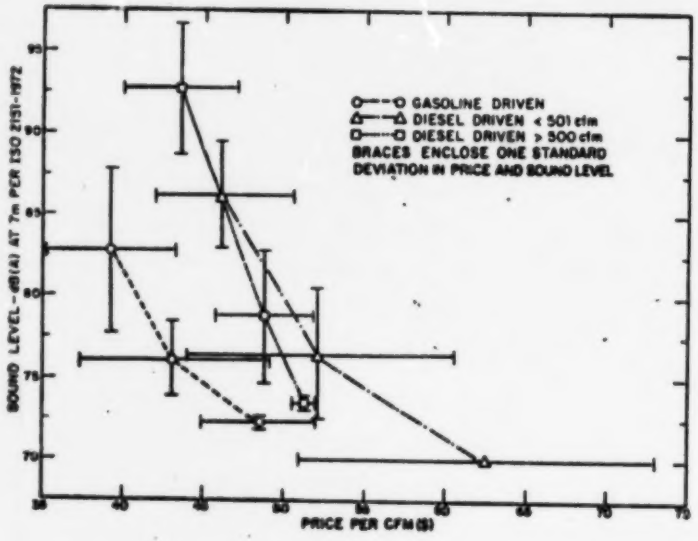

FIG. 7. NOISE/PRICE CONTOURS CORRESPONDING TO CURRENT TECHNOLOGY.

noise reduction that, by our own experience, should not cost the indicated price differential; we have observed equivalent noise reduction treatments for which considerably different price increments are indicated in the list prices; and we have observed competitive compressors that have almost the same list prices for standard and quieted units but that have apparent differences in costs of quieting. There is also a considerable variation in sound pressure levels of compressors that are labelled "quieted" ("muted", "silenced" etc.) (Appendix D); that is, firms are not selling noise reduction as sound pressure level but rather "quieted" as a category. Our conclusion is that the price of "quieted" units, determined in the market by the price leaders, establishes additional revenue per unit that the manufactuers can expect. Some achieve considerable noise reduction, others

TABLE V. MARGINAL COST OF QUIETING COMPRESSORS BASED ON COSTS OF QUIETING EQUALING AVERAGE LIST PRICE DIFFERENCE

		Gasoline Powered	Diesel Powered	
		All CFM Ratings	Below 501 CFM	Above 500 CFM
Standard to Quieted	Sound Pressure Range [dB(A)]	>76.1	>76.4	>78.7
	Marginal Cost ($/CFM/dB)	0.61	0.61	0.36
Quiet to Quietest	Sound Pressure Range [dB(A)]	76.1–72.3	76.4–70.0	78.7–73.5
	Marginal Cost ($/CFM/dB)	1.35	1.60	0.48
Additional Noise Reduction	Sound Pressure Range [dB(A)]	<72.3	<70.0	<73.5
	Marginal Cost ($/CFM/dB)	2.99	4.20	0.64

less, and some are able to take profits by low-cost solutions while others consume all the incremental revenue in costs to stay in the market. Consequently the *cost* of quieted units will have to be determined by an additional survey and industrial engineering estimates. We can, however, indicate the expected range of marginal cost.

We assume, then, that the incremental cost of quieted units ranges over the standard deviation of the mean price. We can also assume that the cost of the quietest units ranges over their corresponding standard deviation. So, since it is unreasonable that the marginal cost should decrease as more quieting is achieved, any noise cost curve must be concave. These considerations lead us to depict the range of noise cost curves as shown in Fig. 8. These data are derived primarily from Table III and correspond to three types of technology: (1) present commonly used sound insulation, (2) present exceptionally used sophisticated sound insulation, and (3) noise source elimination. The marginal costs for "additional noise reduction" are projections obtained by assuming that the margional costs form a geometrical sequence. The marginal costs for the first two ranges are obtained by dividing the references between

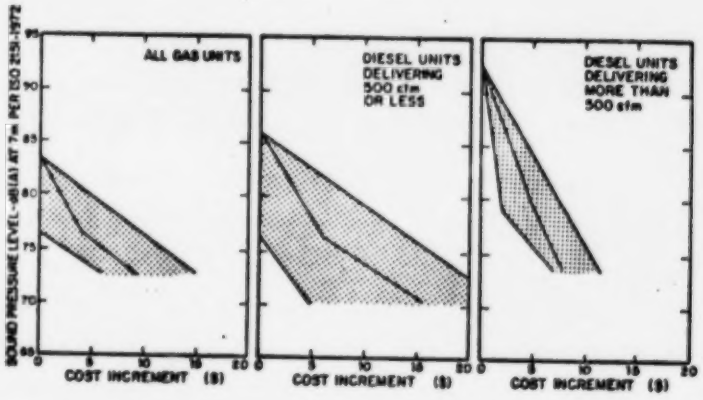
ALL GAS UNITS
DIESEL UNITS DELIVERING 500 cfm OR LESS
DIESEL UNITS DELIVERING MORE THAN 500 cfm
COST INCREMENT ($)
COST INCREMENT ($)
COST INCREMENT ($)

FIG. 8. NOISE/COST CONTOURS FOR CURRENT TECHNOLOGY BASED ON INCREMENTAL COSTS AT 60% OF PRICE DIFFERENTIATION.

average prices of standard, quieted and quietest compressors, a difference between averages of the sound pressure levels for the corresponding categories. For each category there is a lower bound, a mean curve, and an upper bound. The mean curve is that of Fig. 7; it represents cost increments equal to prices. The lower and upper bounds are obtained by using data points corresponding to one standard deviation and using the constraints of negative slope and increasing marginal cost.

These results can be used to project prices for compressor noise control. Using the marginal prices of Table V, we will consider a set of three reduction goals. For the first, we choose sound pressure levels that about half the presently offered quieted compressors already achieve. For the second, we choose levels that 10% of the presently offered machines achieve. For the third, we choose a level close to that achieved experimentally (see Sec. 3.2). These levels (rounded off) and the corresponding prices per cfm appear in Table VI. The prices of Table VI are estimated by multiplying the appropriate marginal prices of Table V into the noise reduction required to achieve each successive level. We begin with the levels and prices of the quieted compressor as given in Table III.

Because these prices are based on current technology, the prices of reduction to level 3 may be high. The compressor industry would be ill-advised to continue noise reduction using the current technology of acoustic insulation. It will not generally achieve the reduction needed to meet the most stringent projected noise limits, and it has reached its economic limits (very steep marginal costs). At present, a 70 to 73 dB noise limit on all compressors should not be a burdening limit since it

TABLE VI. PROJECTED PRICES FOR THREE SOUND PRESSURE LEVELS USING CURRENT TECHNOLOGY

	Gas	Diesel <50l	Diesel >500
Level 1 [dB(A)]*	76	76	78
Price ($/CFM)	43.38	52.35	48.95
Level 2 [dB(A)]*	73	70	73
Price ($/CFM)	47.43	61.95	51.35
Level 3 [dB(A)]†	65	65	65
Price ($/CFM)	71.35	82.95	56.47

*Levels constitute averages of manufacturers reported sound levels that characterize their quieted models. See Appendix C for further analysis.

†Level 3 is an extrapolated level from the averages in Levels 1 and 2.

corresponds to levels already achieved at reasonable costs by some firms. Furthermore, the dropping of the standard lines should produce economies of scale. Ultimately a 65 dB(A) level should be achievable without such a price penalty.

Since development cycles of from less than 1 yr to a little more than 2 were reported by our interviewees we suggest that 18 months is a reasonable time to achieve Level 2. It is our judgment that models that require more than 6 months to quiet and 6 months to put into production cannot be quieted cost-effectively, using current technology. Such models should be removed from the market or redesigned from the bottom up to reach Level 3 in a 3-yr development cycle.

3.2 Quieting by Eliminating Noise Sources

Presently, portable compressors do not emit sounds characteristic of the compressor type, but rather of the engine and fan noise, as can be seen in Fig. 9. Not only are the peaks

attributable to fan and engine exhaust noise, but the continuous spectrum above 500 Hz is also aerodynamically generated noise – vortex shedding by the fan, accelerating air at inlets, etc. Some idea of the noise generating capability of the latter mechanisms is obtained by noting that driving the cooling fans generally takes 3 to 7 times the power involved in the flow of the cooling air. This power is largely consumed in moving air in ways that do not contribute to the flow. In the close spaces where cooling fans operate, aerodynamic inhomogeneities are converted to acoustic radiation by the impedance discontinuities of radiator engines, cowlings, etc. Since reduction of sound generation would also reduce power requirement, this is also a clear example of the cost *reduction* that can be expected from a more advanced quieting technology. It has also been our experience that under certain circumstances the design necessary to reduce noise generation in exhuast systems increases engine efficiency. Similarly, a diesel

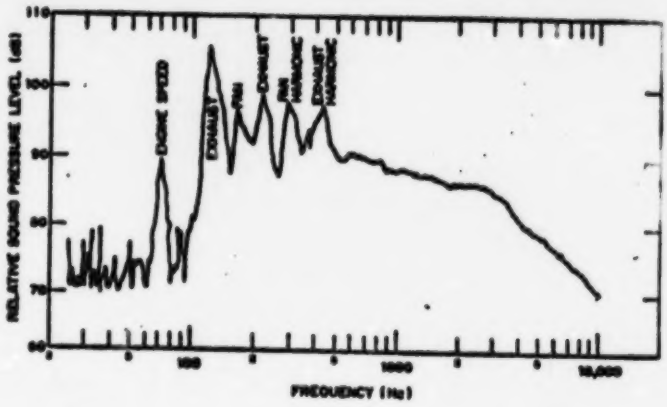

FIG. 9. NARROWBAND NOISE ANALYSIS OF UNSILENCED COMPRESSOR (FROM REF. 4).

with a turbocharger reduces exhaust noise as it increases power capability.

We were privileged to observe performance of a unit* that produced 73.5 dB(A) at 1 m (by our extrapolation this corresponds to 63.2 dB at 7 m). This unit was more efficient than typical units of the same rating. It uses a smaller-than-average engine and had a sophisticated cooling system. Evidence of the sophistication and efficiency of the cooling system design is that opening the side panels decreased the cooling efficiency and caused the machine to overheat so that the thermal cutoff shut it off.

Insufficient data are available to project the economies of noise source elimination. More industry experience is needed with perhaps a well documented pilot project. At present only the marginal costs of Table V and the projected prices associated with Level 3 in Table VI are available. But there is a good technological basis to predict cheaper, quieter compressors if the industry modernizes its noise reduction technology.

3.3 The Influence of Quieting on Operating Costs

Quieting by present technology imposes the penalty of increased operating costs. A larger, heavier compressor must be towed to the job. More fuel is consumed to generate the horsepower to drive an inefficient fan system. Maximum ratings must be adjusted to account for the heat rise in inefficient enclosures.

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*Most of the significant experimental work we observed applying noise source elimination technology was on the smaller gasoline-powered units. This was apparently due to the larger development capital costs involved in experimenting with large expensive compressors. The firms doing the work were small, with less working capital than the industry's leaders in sales.

These fuel penalties have little effect, however, because in the construction industry differential operating costs and capital costs play a minimal role. In practical terms, it is much more costly to a construction operation to have a machine shut down for a few hours than to use additional fuel. However, the construction industry is in flux and shows signs of modernizing. There may soon be considerable pressure on compressor manufacturers to modernize, too. The more efficient compressor design possible with noise source elimination technology will then be in demand. Compressors with efficient ducted cooling systems are also likely to be more reliable.

In summary, present conditions are such that operating cost increases associated with the current technology of compressor quieting are externalities to the compressor industry and of marginal concern to the construction industry. Substantial further noise reduction will bring in new quieting technology which is likely to result in compressors with lower operating costs.

4. RECOMMENDATIONS

Since 50% of the models of "quited" portable air compressors available meet the criterion of 76 – 78 dB(A) at 7 m as measured by ISO 2151-1972, all firms in the industry should be able to meet this requirement at present costs of a "quieted" compressor within 6 months. The smaller firms are generally more advanced in noise reduction technology, so this action is not likely to result in further concentration.

Since 10% of the "quieted" models meet a criterion of 70 – 73 dB(A), and are offered at prices that average $5.79/cfm above the average, the industry ought to be able to meet the requirement of 70 – 73 dB(A) at an average price rise of $5.79/cfm. Manufacturers can achieve this goal within 18 months, and their changes may result in a significant shift in relative market shares.

The present measurement schedule of ISO 2151-1972 does not provide a measurement of noise radiated to upper stories of city buildings. A sound measurement that provides for noise radiated from the top of the machine should be included in a code-related measurement standard.

Present noise reduction technology is effective only when the doors of the compressor enclosure are properly shut. Manufacturers should be required to provide means to compel the user to operate with noise-reduction fixtures in full use.

Portable air compressors are being developed that will meet the criterion of 60 dB(A) at 7 m as measured by ISO 2151-1972. These compressors should ultimately be more efficient with lower operating costs. Firms should be able to produce such compressors to sell for less than current models after amortization of development costs. Three years is the maximum time it should take for

firms to achieve this technology. If the technology is not shared, some firms may not be able to compete and new firms may enter the field. A requirement of 65 dB(A) at 7 m re ISO 2151-1972 in 3 - 5 yrs is therefore a reasonable requirement on the industry.

Imposition of noise limits on engine, fan, and cooling system manufacturers would quicken the response of the compressor industry.

Phasing out unquieted standard models from the market will ultimately result in significant economies of scale as the industry offers only a full line of quiet compressors.

REFERENCES

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2. "Pumps and Compressors 1972," *Current Industrial Reports*, U.S. Department of Commerce Series MA-35P(72)-1, June 1973.

3. *Portable Compressors with Air Delivery Ratings Over 100 cfm at 100 psi*, McGraw-Hill Inc., New York, New York, 1972.

4. D. C. Hinck and W. A. McGahan, "Sound Reduction of Large Portable Air Compressors," Ingersoll-Rand Research Inc.

APPENDIX A

LIST OF CONTACTS

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201-696-0554
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Person Contacted: Brendan Wilkinson (Engineer)

Compair Canada
2185 N. Sheridan Way; Clerkson, Ont., Canada
416-822-7350
Person Contacted: L. King (Vice President, Sales)

Davey Compressor Co.
600 Franklin Ave.; Kent, Ohio 44240
216-673-3471
Person Contacted: Terry Caffrey (Chief Engineer)

Gardner-Denver
Gardner Expressway; Quincy, Ill.
217-222-5400
Person Contacted: Bill Lahrman (Product Engineer)

Ingersoll-Rand Co.
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LIST OF CONTACTS (Cont.)

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Kellogg-American, Inc.
565 Cedar Way, Oakmont, Pa. 15139
412-828-7700
Person Contacted: Mr. Dowling (Stationary Compressors only)

Kent Air Tool Co. (Subs. C.L. Gougler Machine)
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216-673-5821
Person Contacted: Sam Gunning (Imports Compressors)

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Person Contacted: Harish Shah (Manager of Engineering)

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219-879-8211
Person Contacted: Austin Wilkins (Manager, Sales Engineering)

Worthington CEF Inc.
Holyoke, Mass. 01040
413-536-0600
Person Contacted: Loree Paulson (Managing Engineer)

Compressed Air & Gas Institute
Committee on Sound
c/o G.M. Diehl
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201-859-2121
Person Contacted: George M. Diehl (Representative for Sound)

APPENDIX B

EXCERPTS FROM THE MEASUREMENT STANDARD FOR PORTABLE COMPRESSORS ISO 2151-1972 ANCI S5.1-1971 CAGI/PNEUROP

6.0 Procedure for Sound Level Measurement for Portable Compressors

6.1 Operation of Equipment

The machine shall be operated as for continuous service and at its normal rated working pressure.

Idling Condition

The machine shall be running in the idling condition with the discharge valves from the receiver closed.

Full Speed on Load Conditions

The machine shall be running at the design full speed with the compressor on load, delivering its rated output and pressure. The discharge should be piped clear of the test area or fed into an effective silencer.

6.2 Test Environment

This method calls for an environment which can be obtained only on an open site above a hard reflecting plane. An example of a suitable test site would be an open space of which the central twenty (20) meters are covered with concrete, asphalt, or similar hard material.

Care should be taken that gusts of wind do not distort the results of measurements. A microphone windscreen should be used if it is essential to carry out tests in gusty conditions. A calibration correction shall be applied.

6.3 Background Sound Level

The background sound level, when the machine on test is not running, shall be determined at one of the microphone locations to be used when conducting the test. The readings at each location, with the machine running, should exceed the background levels by at least ten (10) dB in each octave band of interest. When the difference is less, corrections should be applied as in Appendix I. If the difference between the measured sound and the background sound in any octave band is less than three (3) dB, a valid sound measurement of the machine cannot be made in that octave band.

Octave bands of interest are defined as those where the sound pressure levels produced by the machine are within forty (40) dB of the highest measured octave band level, but in no case are levels below fifty (50) dB considered important.

6.4 Sound Measurement

The following microphone locations shall be used for the test and the prescribed measurements shall be taken:

6.4.1 Microphone Locations

A preliminary survey shall be taken all around the machine at a distance of one (1) meter from the nearest major surface and at a height of 1.5 meters to establish the direction of the maximum overall sound level (A scale). The microphone locations shall be in this direction and in four other directions located at each end and at the center of the sides of the machine.

Two sets of microphone locations shall be used (see Fig. B.1):

(a) One (1) meter distance from the nearest major surface of the machine and at a height of 1.5 meters above the reflecting plane.

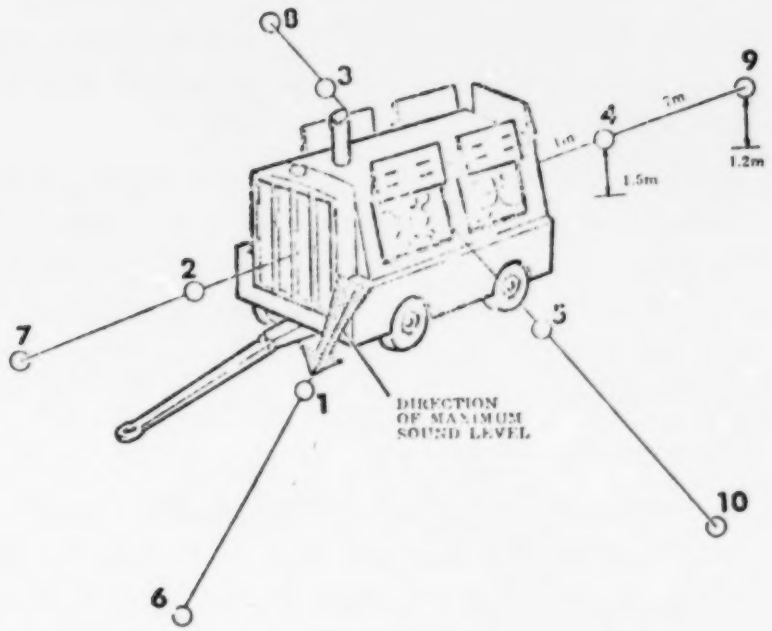

FIG. D.1. PORTABLE COMPRESSOR.

(b) Seven (7) meters distance from the nearest major surface of the machine and at a height of 1.2 (1.5m in ISO 2151-1972) meters above the reflecting plane.

6.4.2 Measurement Technique

Because of the interference between direct sound waves and those reflected from the ground, large errors may occur when strong discrete frequency components are present. Tests shall be made by rapidly (minimum one (1) cycle per second) moving the microphone vertically approximately ±0.3 meter from each location. The microphone shall be held in the position, grazing incidence or perpendicular incidence, in which it was calibrated for flat response.

The period of time during which the measurements are made shall be long enough to allow an average reading to be taken with the slow response setting of the meter.

No reflecting surfaces shall be near the microphone. Observers and measuring instruments shall be at least one (1) meter away from the microphone and the machine under test (microphone must not be placed between observer and machine). Precautions shall be taken to minimize interference of the sound measurement by the operating personnel.

6.4.3 Measurements to be Taken

The following measurements are to be taken: first, at one of the microphone locations for background sound without the machine operating; second, in each of the microphone locations with the machine operating under the conditions of paragraph 6.1:

(a) Overall sound level using the "A" weighting network.
(b) Octave band sound pressure levels using the flat response network.

8.0 Calculation and Interpretation of Readings

8.1 Calculations

Readings should be corrected for background sound. Calibration corrections, if required, shall be taken into account. The average of the corrected sound pressure level readings at the specified microphone locations may be calculated according to the following rules:

8.1.1 Maximum Variation Five (5) dB or Less

If the maximum variation in corrected sound pressure levels is five (5) dB or less, average the sound pressure level values arithmetically.

8.1.2 Maximum Variation Five (5) dB to Ten (10) dB

If the maximum variation in corrected sound pressure levels is between five (5) dB and ten (10) dB, average the sound pressure level values arithmetically and add one (1) dB.

8.1.3 Maximum Variation Over Ten (10) dB

If the maximum variation exceeds ten (10) dB, average according to the equation below:

$$[L] = 10 \log_{10} \frac{1}{n} \left[\text{antilog}_{10} \frac{L_1}{10} + \text{antilog}_{10} \frac{L_2}{10} \text{ --- } + \text{antilog}_{10} \frac{L_n}{10} \right]$$

where [L] = Average sound level (dB A) (or band average pressure level in decibels)

L_1 = Sound level (dB A) (or band sound pressure level in decibels) at first position

L_n = Sound level (dB A) (or band sound pressure level in decibels) at nth position

n = Number of measuring positions.

8.2 Presentation of Data

A test report shall be supplied and shall give the following information:

8.2.1 Statement that the test was conducted in accordance with the CAGI-PNEUROP Test Code.

8.2.2 Description of the machine, operating conditions, and a sketch showing the test layout and microphone locations.

8.2.3 Make, model, and serial number of instruments used.

8.2.4 Tabulation of test data:

8.2.4.1 Corrected sound level measurements at each measuring point (dB A), and corrected octave band sound pressure levels.

8.2.4.2 Background sound level at one (1) point (dB A), and octave band sound pressure levels.

8.2.4.3 The average of the corrected (dB A), and octave band sound pressure levels (optional).

8.2.5 Plotting of test data: Octave band data shall be plotted for the microphone location at which the highest sound level (dB A) was recorded.

SOUND LEVEL TEST REPORT FOR PORTABLE COMPRESSORS PER CAGI-PNEUROP CODE

REPORT FORM NO. 8.3.2

SUBJECT:

Model: _____ Manufacturer: _____ Serial: _____

Rated Speed & Capacity: _____

Description: _____

_____

TEST CONDITIONS:

Distance from Subject to Microphone: _____ Meters

Operating Speed as Tested: _____ Air Pressure Supplied: _____ Bars

Height of Microphone Above Reflecting Plane: _____ Meters

Reflecting Plane Composition: _____

Remarks: _____

_____

INSTRUMENTATION:

Microphone: _____ No. _____

Sound Level Meter: _____ No. _____

Octave Band Analyser: _____ No. _____

Calibrator: _____ No. _____

Other: _____ No. _____

DATA:

dB Ref. 2×10^{-5} N/m²		BACK-GROUND	IDLE CONDITION LOCATION •					AV.	FULL LOAD CONDITION LOCATION •					AV.
			1	2	3	4	5		1	2	3	4	5	
MID BAND FREQ. H.	dB A													
	63													
	125													
	250													
	500													
	1 k													
	2 k													
	4 k													
	8 k													

•Corrected for background sound. Readings having 3 dB corrections must be reported in brackets. Only octave bands of interest per paragraph 6.3 need to be reported.

TESTED BY: _____ DATE: _____

REPORTED BY: _____ DATE: _____

Sketch showing microphone locations, orientation of the subject, (direction of exhaust and size of load cell where applicable).

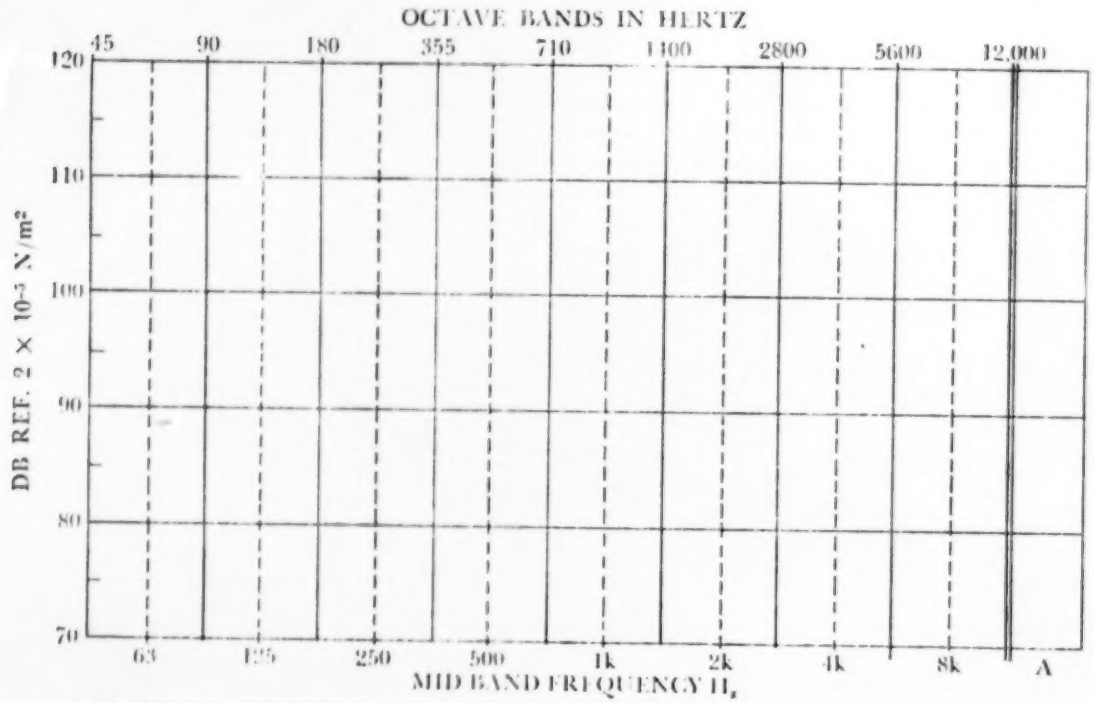

DIAGRAM SHOWING AVERAGE SOUND LEVELS AND BACKGROUND SOUND LEVELS AT TIME OF TEST

APPENDIC C: EFFECTS OF ADDING A MANUFACTURER'S SOUND LEVEL TOLERANCE

When this report was originally developed, the levels of quieting were not defined for purposes of regulation. The average price and corresponding average sound levels were presented in terms of compressor categories and sizes. No provision was available to determine the accuracy of the manufacturer's supplied data or whether the information supplied represented specific machines or characterized models. In developing sound level limits for regulation, we considered the data supplied by manufacturers to be typical or average for the standard models and characteristic or advertised for the quieted versions. That is, the standard models are not advertised for sound level or complying with certain noise codes while the quieted versions are. We therefore projected customer price increases to quiet air compressors on a not-to-exceed basis using manufacturer's supplied data for the quieted compressors.

After the economic impact contractor had contacted the manufacturers, claims were made that the sound level information provided on quieted air compressors was the average or typical value for some of the reported units. Discussions with manufacturers on the normal variance in sound level between identical models indicated that a 2 dB(A) manufacturing tolerance was needed for a not-to-exceed sound level limit. The 2 dB(A) tolerance was then incorporated into the average customer price increase for quieted compressors in the following manner.

For each of the three categories of air compressors, three levels of quieting were selected. Level 1 corresponds to the mean or average sound level of the quieted models reported. Level 2 corresponds to the mean of the lowest decile or the

quietest of the quieted. Level 3 corresponds to the best demonstrated technology in one category (gasoline driven). Since the best demonstrated technology was for a prototype model not available for sale, no customer list price was provided. To project the cost of quieting to the best technology level [65 dB(A)], we used the available information for Levels 1 and 2. If we plot the three points on a sound level *vs* cost graph corresponding to the mean sound level and price for standard (A), quieted (B), and quietest (C) as in Fig. C.1, two lines (AB, BC) may be drawn connecting the three points. The slope of these lines are the average marginal costs of quieting in $/cfm/dB(A) from one point to another. We now add another line to point C whose slope corresponds to a geometric progression in the marginal cost of quieting. The intersection f this new line with the 65 dB(A) sound level forms point D, the projected cost for a best technology compressor based on a geometric progression of costs.

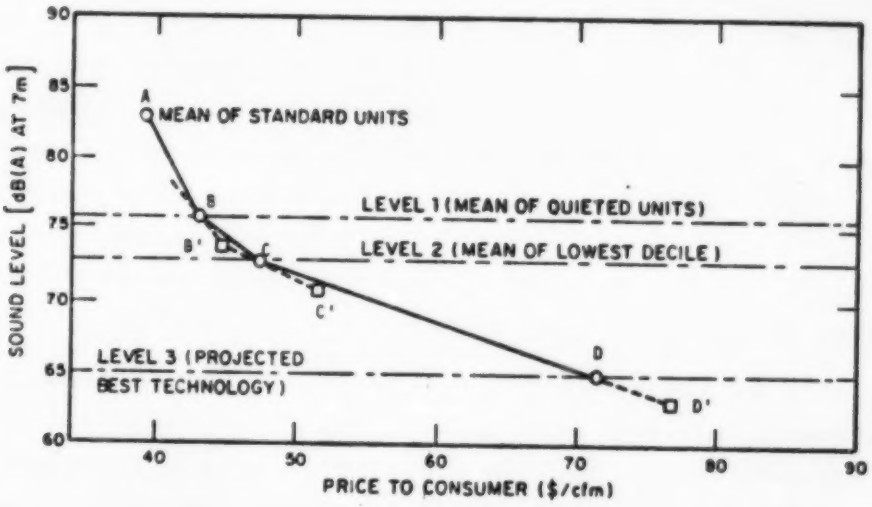

FIG. C.1. EXAMPLE FOR DEVELOPING APPROPRIATE COSTS FOR MANUFACTURERS' SOUND LEVEL TOLERANCE.

All through the previous discussion on points and connecting lines, we must not lose sight of the statistical nature of these points and the real significance of the connecting lines. Each point is an average value both in sound level and price. These points presumably lie on a smooth curve through all the average values of sound level and price combinations. The two lines connecting the points represent chords of the smooth curve. The third line is developed from the assumption that the increased marginal costs of quieting form a geometric progression.

To add in a cost increment to allow provision for a 2 dB(A) manufacturer's tolerance, we ideally must locate the corresponding intersection of the smooth curve of average prices with the sound level 2 dB(A) below the limit. Since the curve is not developed completely, we must use the chord approximations discussed earlier. In providing the greatest degree of accuracy in projecting price increments for a manufacturer's tolerance, the marginal cost of quieting in the neighborhood of the points should be used. At points B and C, the marginal cost of quieting [$/cfm/dB(A)] is the derivative of the sound level *vs* cost curve at these points. This is most accurately predicted, using the points and lines available, on the average of the marginal cost of quieting on both sides of the respective points. For example, looking at point B of Fig. C.1, from Table V on page 22, the slope of line AB (the marginal cost of quieting) is $0.61/cfm/dB(A). Line BC has a slope of $1.35/cfm/dB(A). In the neighborhood of point B, the marginal cost of quieting is the average of these values or $0.98/cfm/dB(A). The increased customer cost for a manufacturer's tolerance is 2 dB(A) times the neighborhood marginal cost of quieting (noted in Fig. C.1 as B' and C'). Since point D is a projected point, we have merely extended the line to D' for the manufacturer's allowance.

Table C.1 lists the average and projected prices with allowance for sound level deviation for the three air compressor categories.

TABLE C.1. CUSTOMER PRICES FOR AIR COMPRESSORS WITH TOLERANCE ALLOWANCE

	Gasoline Driven	Diesel Driven < 501 cfm	Diesel Driven > 500 cfm
Level 1 [dB(A)]	76	76	78
Price ($/cfm) average	43.38	52.35	48.95
Add for 2 dB(A) tolerance	1.96	2.21	.84
Price ($/cfm) not-to-exceed	45.34	54.56	49.79
Level 2 [dB(A)]	73	70	73
Price ($/cfm) average	47.43	61.95	51.35
Add for 2 dB(A) tolerance	4.34	5.80	1.12
Price ($/cfm) not-to-exceed	51.77	67.75	52.47
Level 3 [dB(A)]	65	65	65
Price ($/cfm) projected	71.35	82.95	56.47
Add for 2 dB(A) tolerance	5.98	8.40	1.28
Price ($/cfm) not-to-exceed	77.33	91.35	57.75

APPENDIX D. SOUND LEVEL AND PRICE DISTRIBUTION FOR PORTABLE AIR COMPRESSORS

After completing the initial draft of this report, numerous questions were asked as to the sound levels and prices considered typical for portable air compressors. To provide this information, two graphs were developed plotting sound level vs. capacity (size) (Fig. D1) and list price vs. capacity (Fig. D2). These figures are based on manufacturers submitted data and not all points have been plotted due to overlapping.

The most notable feature of Fig. D1 is the wide variance in sound levels for compressors. Some standard units are quieter than corresponding "quieted" units. Other than a general break or jump at 500 cfm, the sound levels do not display a relationship to the size of the machine. No compressors are reported in the 500-599 cfm range. No gasoline compressors are reported larger than 250 cfm. Using this figure for comparison, we conclude that the categories selected for this report (gasoline driven, diesel driven <501 cfm, diesel driven >500 cfm) exhibit similar sound generating characteristics within the respective category.

Figure D2 is plotted on log-log coordinates scale to compress the range of prices and capacities at the high end and still provide some clarity for the prices of the smaller units. Using the log scale, the relative distance between data points for quieted and standard units over the range of capacity corresponds to the percentage difference in price. Thus while the price increase for quieting a 1000 cfm machine is $10,000, the price increase for the 100 cfm machines is $1,000 for equal percentage increases based on capacity. The range of prices vary greater in the smaller categories than in the larger one. This

is also evidenced in Fig. 7 on page 21 where the price deviation noted for the diesel compressors larger than 500 cfm is less than the gasoline or small diesels.

We conclude than, that the compressor categories selected are adequate and the normalizing factor used for price ($/cfm) is reasonable as Fig. D2 shows somewhat of a straight line in ($/cfm) over the three categories.

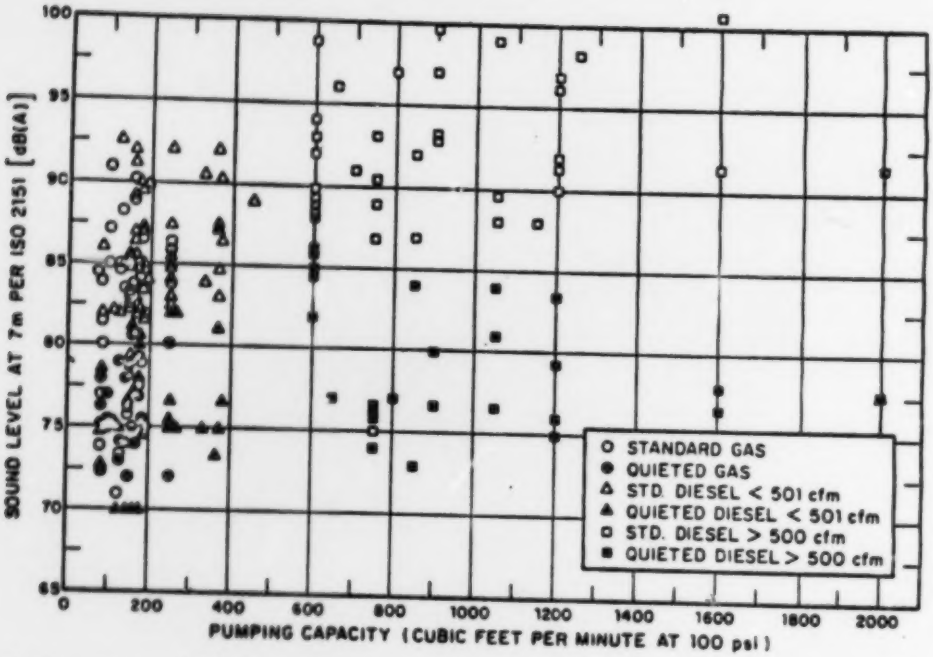

FIG. D1. SOUND LEVELS OF PORTABLE AIR COMPRESSORS AS A FUNCTION OF PUMPING CAPACITY

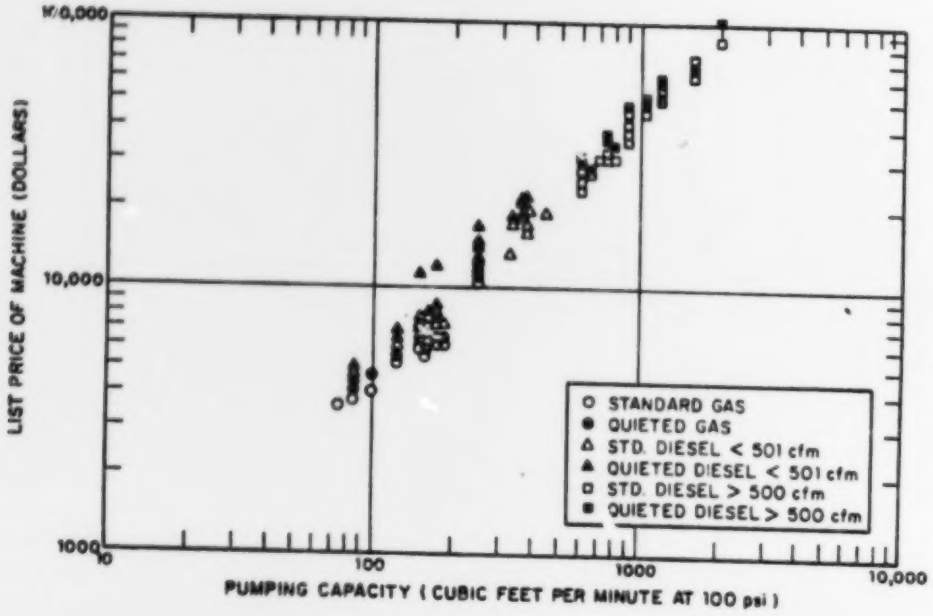

FIG. D2. PRICE OF PORTABLE AIR COMPRESSORS AS A FUNCTION OF PUMPING CAPACITY